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Cylinder-head surface temperature and maximum pressure in a diesel engine

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CYLINDER-HEAD SURFACE TEMPERATURE
AND
MAXIMUM PRESSURE
IN A
DIESEL ENGINE

by

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A

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ABSTRACT

This report presents the result of an experimental investigation on the effect of 1) fuel consumption in a diesel engine on the mean cylinder-head surface temperature 2) engine speed and load on the maximum cylinder pressure.

A fast-response thermocouple was fixed in the cylinder head and temperatures were measured for various load and speed conditions. An attempt was made to record a transient surface temperature versus crank-angle diagram. Some doubt as to the reliability of this recording may arise because of the very small amount of time lag in the thermocouple action. This is immaterial to the subject of this investigation since we are dealing here with mean temperature only.

A pressure transducer was used to measure cylinder pressure. Signals from this pressure transducer and crank-angle signals from a magnetic pickup were transmitted to an oscilloscope and the recording of the pressure versus crank-angle diagram was obtained by means of a drum camera. Maximum pressure and crank angle at maximum pressure were measured for various load and speed conditions.

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The writer wishes to express his gratitude to Dr. A. Feingold for suggesting this problem and for his guidance throughout the investigation and to Dr. A. J. Miles for supporting this project.

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Last, but not least, the complicated instrumentation could never have been completed in time, were it not for the patience, devotion and capability of Messrs. Lee Anderson and Richard Smith of the Mechanical Engineering Department. Mr. Arthur R. Hemme, Laboratory Technician, did part of the necessary machining.

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I. INTRODUCTION

The origin of the internal combustion engine dates back to 1886, when development work was undertaken independently by Herbert Akroyd Steward in England and Rudolf Diesel in Germany (1).

Dr. Rudolf Diesel established the first patent in 1892, which was to use powdered coal as fuel. This first engine was not successful due to the difficulties in proper atomization of the fuel but nevertheless the theory underlying its operation was practicable. Since then, many modifications and improvements took place before a modern diesel engine came into existance.

A typical modern diesel engine power plant with a high compression ratio and supercharger possesses the highest thermal efficiency ever achieved by a power-generating cycle working in similar thermodynamic conditions (2). For this reason, the diesel engine may be considered as the best power plant available at present for ships, railroads and for heavy duty commercial vehicles.

It is believed that the gas turbine due to its simplicity and the nuclear power plant due to its lower running cost will be competitors of the diesel engine in the future (2). Especially for small units as used in automobiles or for medium-power units as may be used in industries, gas turbines have already been designed and are now being

tested on test-benches and in actual operations. Keeping this in mind, more and more efforts are being made at present to improve the thermal efficiency and the performance of the diesel engine without adding undue complexity either in manufacturing or in its operation.

Much basic research has been done to find out what is happening inside the cylinder, to learn about pressure and temperature variations, detonation, fuel injection and combustion etc. Still, there are many theoretical and practical problems unsolved due to inadequate or insufficient data. Experimental work carried out in the past was sometimes either questionable or inconclusive because the equipment available did not give sufficiently accurate measurements.

In recent years, means of measuring thermodynamic and mechanical quantities with high accuracy have been developed. It is hoped that more intensive experimental work will be carried out and will be utilized for still more rational designs of diesel engines.

II. REVIEW OF LITERATURE

From the design and research point of view, it is essential to have precise knowledge of thermodynamic quantities in the cylinder.

Measurements of gas temperature can be accomplished by either pyrometer or a resistance thermometer or by sound-velocity method. All these methods have been described in great detail in the literature (3, 4, and 5). As the gas temperature varies during the cycle, cylinder-head temperature varies as well, but not significantly. The magnitude, both of the wall temperature and of its variation, depends upon the location.

In 1939, Eichelberg (6) had measured cylinder-surface temperature at different depths of wall thickness and concluded that variation of temperature during the cycle decreases with the distance from the inner surface.

Judge (7) and Zanoni (8) have described in some detail the special features of thermocouples for measuring transient temperatures. However, there is some doubt about the ability of any thermocouple to measure the very rapid variations of surface temperature in high speed diesel engines, because the time lag can be minimized, but not completely eliminated. In a detailed study of the heat transfer through the cylinder head we may be interested in the variations of the surface temperature, but for the

purpose of the present investigation we shall consider the mean temperature only. Gross-Gronomski (10) has found a simple relationship between relative piston temperature and fuel consumption.

There are several types of pressure transducers available which give some kind of electrical output when pressure is applied to a diaphragm. These include condenser-type, piezo-electric type and strain-guage type pressure transducers. These and other pressure measuring devices are discussed in detail in the literature (7, 10, and 11).

III. EXPERIMENTAL TECHNIQUE

General Apparatus and Test Procedure

A view of the general setup used in this investigation is shown in Figure 1. The various component parts of the apparatus are described in the following section.

Diesel Engine:

The diesel engine was a single cylinder, Nordberg type 4FS. This is a 4-stroke, vertical, cold starting engine. Its specifications are as follows:

Model:	4FS1-AE
Bore:	4 1/2 inches
Stroke:	5 1/4 inches
Piston-Displacement:	83.5 cubic inches
Compression Ratio:	14.6:1
Speed Range:	1200-1800 r.p.m.
Engine Brake Horse Power:	10-15 H. P.

Dynamometer:

The engine was coupled with a Westinghouse multipurpose D. C. dynamometer. Both the engine and dynamometer were provided with a control panel for starting and to control the speed and load, Figure 2. The dynamometer can also work as a motor and was used to start the engine.

Crank Angle Marker (Magnetic Pickup):

To provide a horizontal coordinate for the pressure and temperature display on the oscilloscope, the following

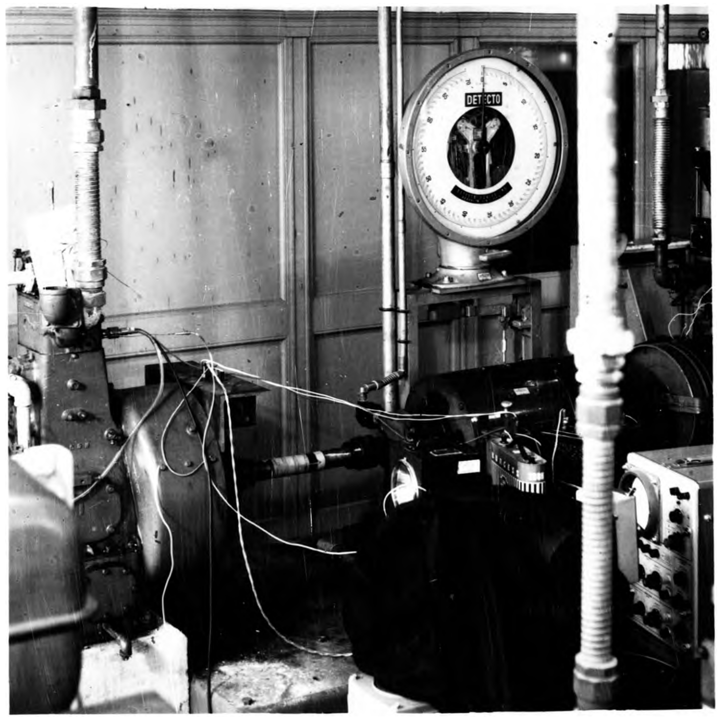


Figure 1. General Laboratory Set-up of Experiments

Figure 2. Control Panel

electro-magnetic arrangement was used. Two coils, a primary and a secondary one, were wound around an iron rod. This rod was chisel-shaped at the lower end and positioned close to the circumference of an iron disc, Figure 3. The disc was keyed on to the engine main-shaft and was carefully checked to ensure the impossibility of relative movement between the shaft and the disc. The circumference of the disc was divided into 360 equal parts, each part corresponded to one degree of crank angle. Slots were cut at each division. Every fifth slot was made deeper than the others and the slots corresponding to top and bottom dead center cut deepest of all.

The primary coil when connected to a storage battery, created a magnetic flux. The chisel-like shape of the rod directed the magnetic flux to a small area on the circumference of the disc. When the engine shaft and the disc rotated, slots of the disc were passing in rapid succession under the chisel. The air-gap between the iron rod and the disc was thus subject to rapid variations causing the magnetic flux to fluctuate. These changes in magnetic flux were inducing a current of variable voltage in the secondary coil. When the secondary coil was connected to the oscilloscope, it produced a trace in the form of a wave. The peaks corresponding to every fifth slot had greater magnitude than the others. The peaks representing T.D.C. and B.D.C. were still more prominent. This difference in magnitude caused

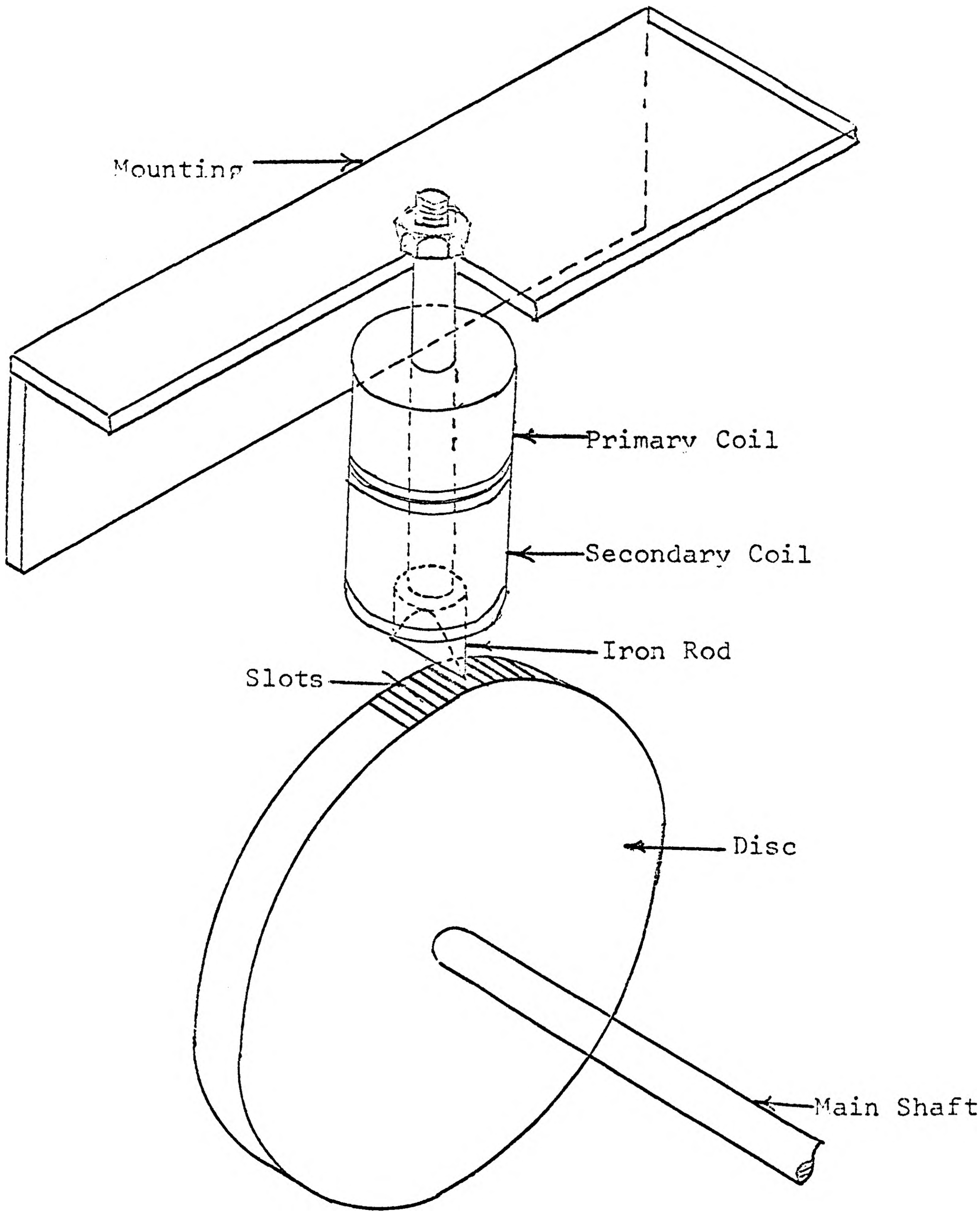


Figure 3. Crank-Angle Marker

Magnetic Pickup

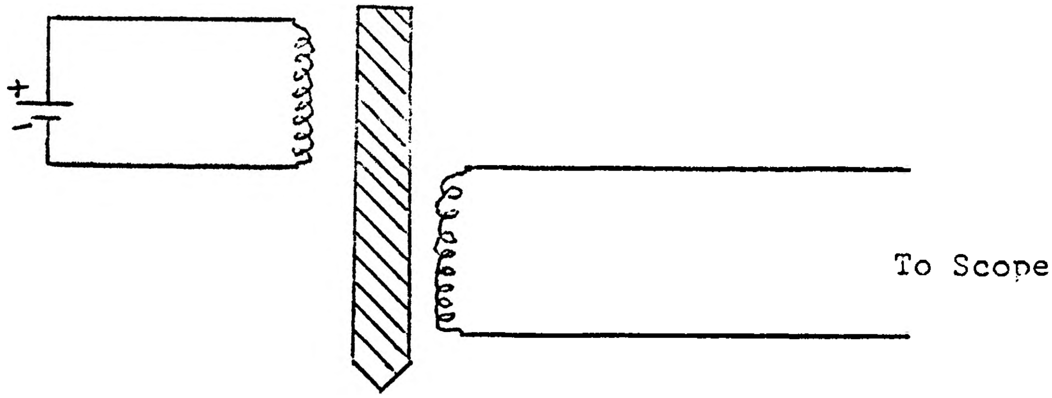
by different depth of slots was helpful in identifying the exact crank angle at every point.

At the beginning, when the circuit was tested, (Figure 4a), it failed to give satisfactory crank-angle signals. Two reasons were discovered. The engine and the pickup mounting were vibrating considerably and, in addition, the connecting wires between the magnetic chisel and the oscilloscope were picking up electric noise from the environment.

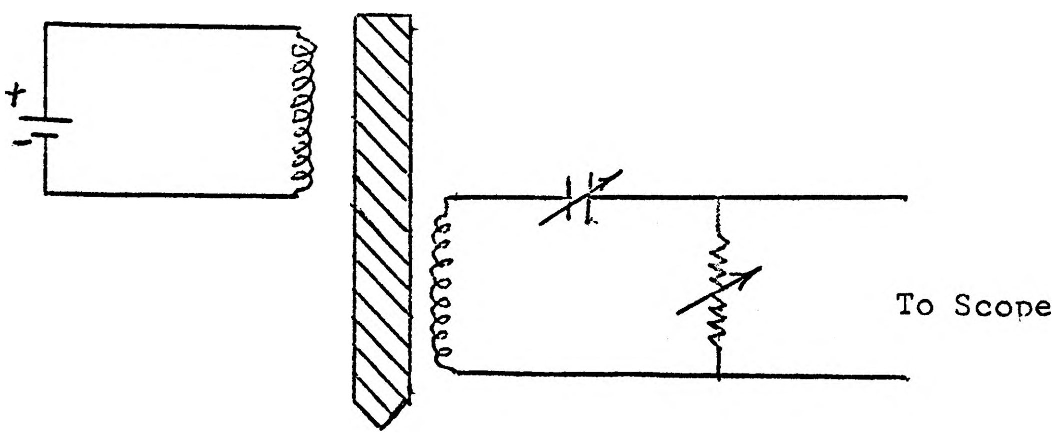
First, the pickup mounting was replaced by much sturdier and a more rigid construction. Then, the connecting wires were exchanged for special low-noise cables. This still failed to provide satisfactory results and the following arrangement had to be incorporated in the circuitry.

The secondary circuit had a variable capacitor and a variable resistor incorporated in it as shown in Figure 4b.

The capacitive reactance $X_c = 1/2\pi fc$ is selective, in the sence that it is large for small frequencies and vice versa. This enables the capacitor to filter out lower frequencies. Actually there will be a certain, particular value of capacitance C for a given value of frequency f for which the current passing through the capacitor at this frequency will be maximum. This will happen when $X_c = X_L$, where the inductive reactance X_L refers to the secondary winding on the chisel. It is so because



(a) Simple Circuit



(b) Capacitance and Resistance
as a Filtering Unit

Figure 4. Development of Crank-Angle Marker Circuit

$$I = \frac{V}{\sqrt{[R^2 + (X_c - X_L)^2]}}$$

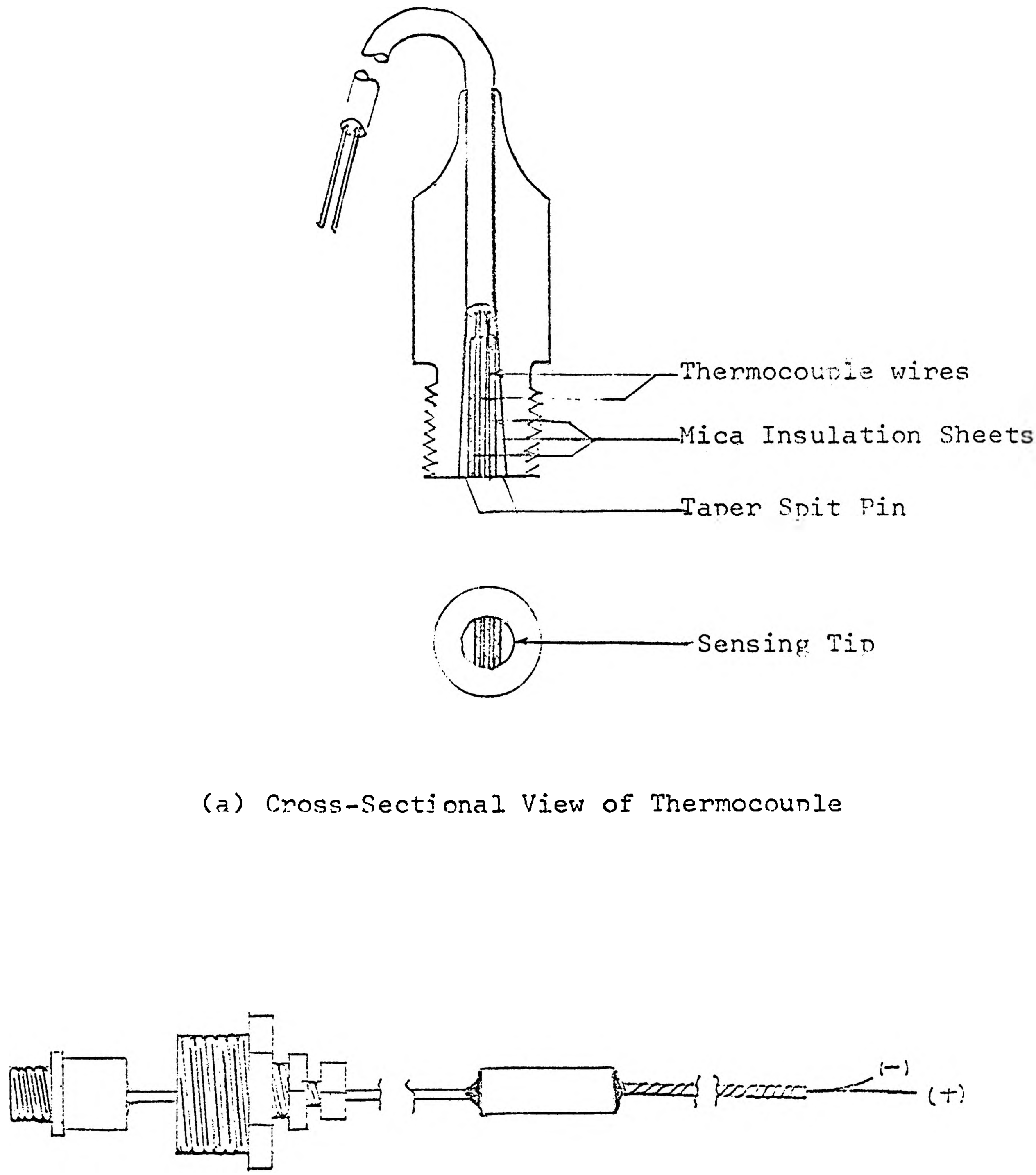
The resistance R of the variable resistor has to be adjusted to obtain minimum noise interference. If we plot current versus frequency, the shape of the plot will be affected by the value of R and therefore the particular noise frequencies which we wish to eliminate, will be suppressed to a greater or lesser degree, according to the choice of R.

After this improvement was introduced, the noise was almost entirely filtered out, as can be seen in Figure 10.

Thermocouple:

In this investigation an eroding-type thermocouple manufactured by Nanmac Corporation, was used. Eroding-type thermocouples are specially designed to measure surface temperatures. Figure 5a shows cross-sectional and end-views of such a thermocouple. These are made in the following manner. Thermocouple wires are flattened and insulated from each other and from the outer body by thin mica sheets. The whole assembly is then pressed into a thermocouple hole at high pressure. A hot junction is formed by grinding and polishing across the sensing tip. At the high temperature in the presence of harmful gases this tip might be subject to erosive or abrasive action but this action simply forms a new hot junction while the sensing tip is wearing away.

The thermocouple was provided with pressure sealing



(a) Cross-Sectional View of Thermocouple

(b) Thermocouple and its Sealing Glands

Figure 5. Thermocouple

glands, Figure 5b. It was fixed in the cylinder head with the sensing tip kept flush with the inner wall-surface. A cross-section of the cylinder head and location of the thermocouple is shown in Figure 6.

The specifications of the thermocouple are as follow:

Type:	Modified Model F
Metals of Thermocouple:	Chromel versus Alumel
Temperature Range:	Over 2000°F
Response Time:	Less than 10 micro-seconds
Pressure Range:	Over 50,000 psi
Velocity Range:	Over 6000 fps.

The last two are rather special features, important in this kind of investigation, because inside the cylinder the working medium reaches both high pressure and high velocity during certain parts of the cycle.

The standard calibration tables published by the National Bureau of Standards were used. The cold junction of the thermocouple was not kept in the ice because this arrangement adds more wiring to the thermocouple circuit and creates new noise-filtering problems. The cold junction was kept at room temperature and the observed thermocouple temperature was corrected by adding the difference between the room temperature and 32°F. Error produced by this method can be ignored as it is within the limit of error of the thermocouple.

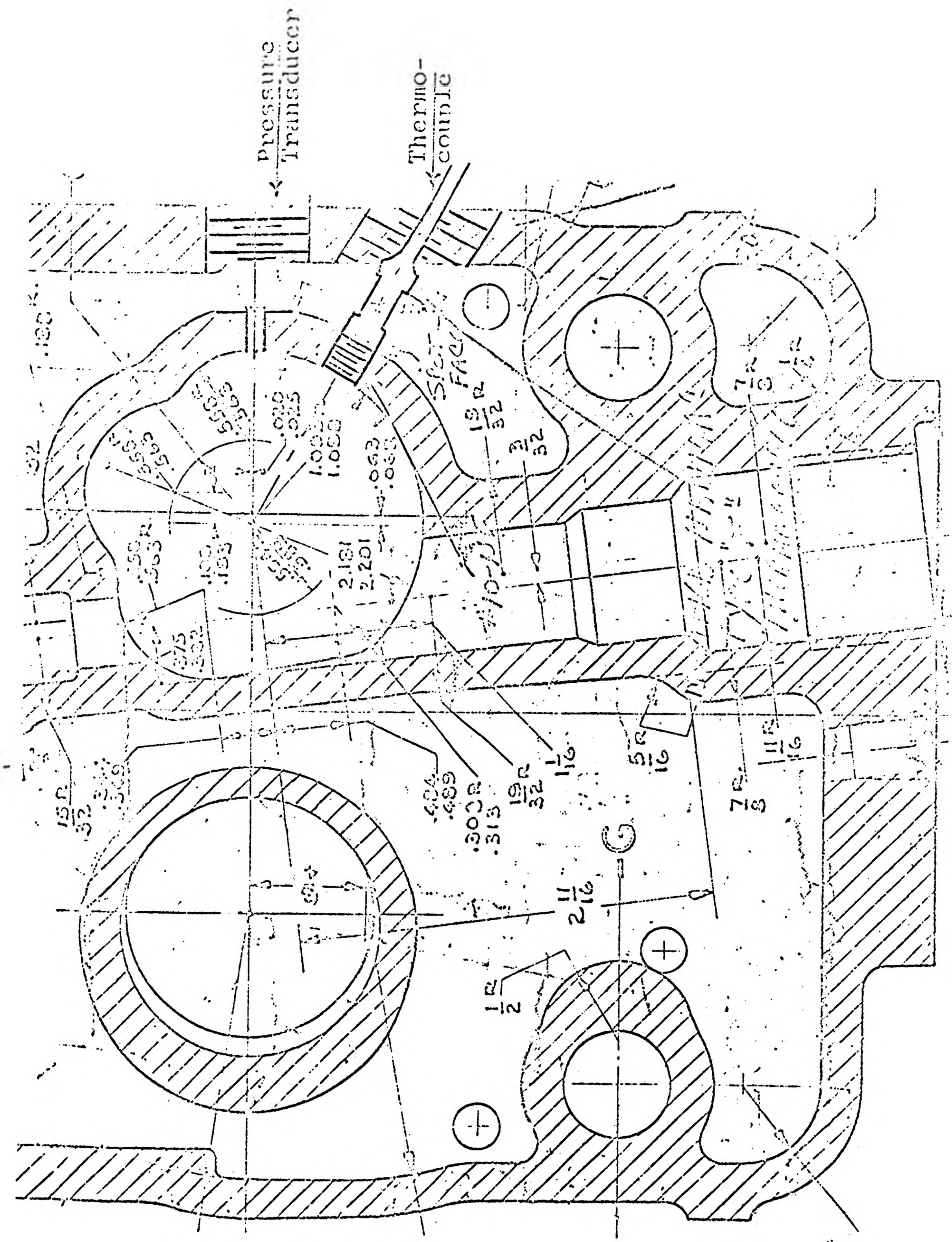


Figure 6. Location of Thermocouple and Pressure Transducer

Pressure Transducer:

A strain-guage type ATL Norwood pressure transducer was used. Brieflv this type of pressure transducer consists of a flush stainless steel diaphragm and a small strain tube covered by circumferential and longitudinal strain-gauge windings, Figure 7. Temperature-compensating winding and a complete bridge circuit are permanently bonded to the strain tube.

Applied pressure causes minute deflection of the diaphragm and compresses the strain cylinder, resulting in an unbalanced resistance in the bridge circuit. The bridge gives an electrical output proportional to the pressure.

Specifications are as follows:

Model No:	110-2-5000-34-10-61
Sensitivity:	3.909 MV/V
Input Resistance:	350.0 ohms
Mounting Torque:	35 ft. lbs.
Excitation:	10 V
Pressure Range:	5000 psig
Coolant Inlet Pressure:	30 psig water
Non Linearity and Hysteresis Combined B.S.L.:	0.60 F.S.

A water-cooling system was provided to keep the temperature of the strain body and windings low. A transducer was fixed in the cylinder head as shown in Figure 6.

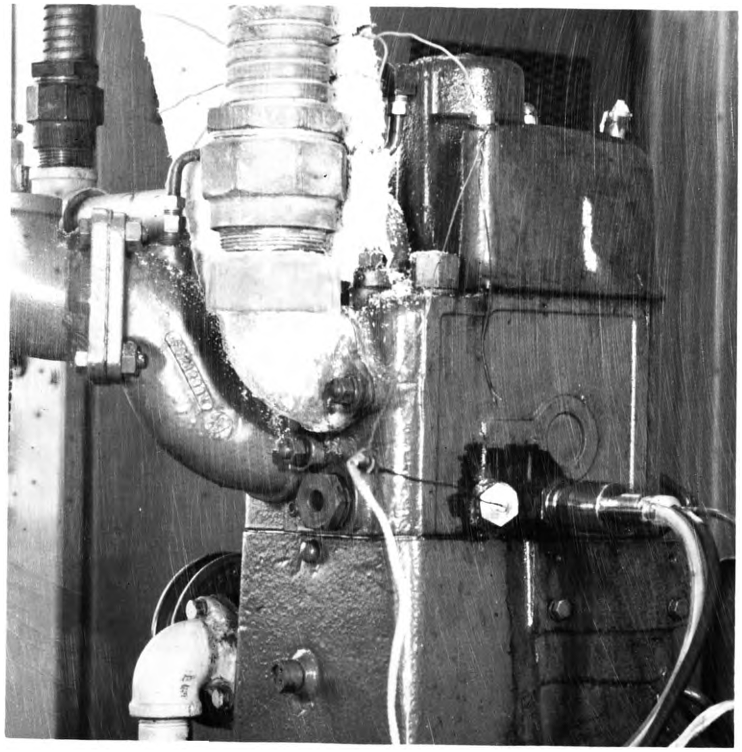


Figure 7. Pressure Transducer

A standard battery was used for excitation. At the end of the test, the transducer was removed from the engine and was calibrated on a dead-weight tester, Figure 8.

Oscilloscope:

Signals from the thermocouple pressure transducer and crank-angle pickup were transmitted to a dual-beam oscilloscope. Originally an Analab Model 1120 was used. This scope has only one electronic gun and the beam is divided in two parts when it is used as a dual-beam scope. Light intensity is thus reduced and does not give a satisfactory impression on the photographic film. For this reason it was found necessary to replace it with a Tektronix Model 502, dual beam differential input oscilloscope.

Drum Camera:

A drum camera was used to record the pressure versus crank-angle diagram and temperature versus crank-angle diagrams. The drum camera constructed for this test consisted of a 10-inch drum completely covered by black cloth and equipped with a lens and a shutter, Figure 9. The film was wrapped around the drum, supported on a specially provided shoulder. The ends of the film were fastened by means of a screw-down wedge. A one-half horse power U.S. Motors vari-drive electrical motor was used to revolve the drum. This motor had a speed range from 103 rpm to 1030 rpm. It was supported on a rigid base to reduce vibration.

Figure 8. Dead-Weight Tester

To take a picture of the oscilloscope the drum was loaded with film and was driven at half the diesel engine speed. Appropriate signals were transmitted to the scope and the shutter was kept open during one complete revolution of drum. The horizontal sweep of the oscilloscope was disconnected during the time of photography, since the rotation of the drum provides horizontal travel.

Because the film passes at a high speed in front of the lens, films of highest available speed and a lens of minimum f value available must be chosen. The lens actually used had a f = 1.9.

Temperature Measurement

During this part of our investigation the thermocouple was fixed on the engine and a series of experiments were performed with four different speeds and four different loads. Several temperature versus crank-angle diagrams were photographed and the mean temperature obtained from them, (Figure 10). For other experiments the thermocouple output was measured with the help of a potentiometer. Fuel consumption was also measured each time.

Pressure Measurement

The engine was run at various loads and speeds. Pressure signals and crank-angle signals were transmitted to the scope and recorded on photographic films. The cool-

ing water flow rate, its inlet and outlet temperature and fuel consumption of the engine were measured by conventional methods.

Calibration of the Pressure Transducer

The pressure transducer was fixed on a dead-weight tester. The pressure was increased from 0 psig to 600 psig by increments of 50 psi and then decreased gradually, again at the rate of 50 psi at a time. The same battery, used during the test, was used for excitation. Output of the pressure transducer was measured by a potentiometer.

Table 1 shows the readings obtained, and Figure 15 shows the calibration curve. This curve does not pass through the origin but shows some initial reading for 0 psig. This initial error might be due to an improper fixing arrangement for the pressure transducer on the dead-weight tester, resulting in a minute compression of the diaphragm. A corrected curve parallel to the original curve and passing through the origin, was drawn, Figure 15.

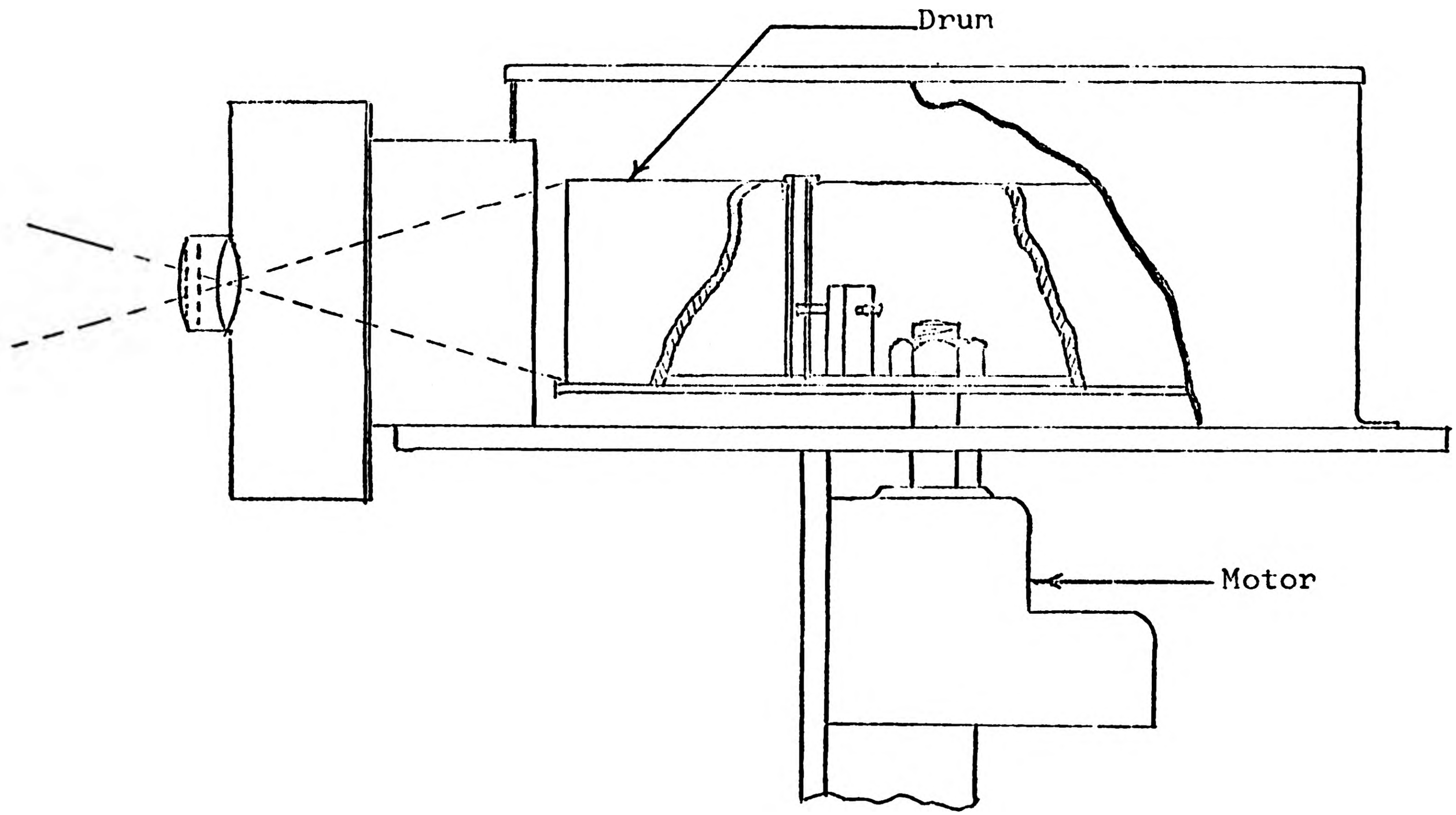


Figure 9. Drum Camera (Excluding Black Cloth)

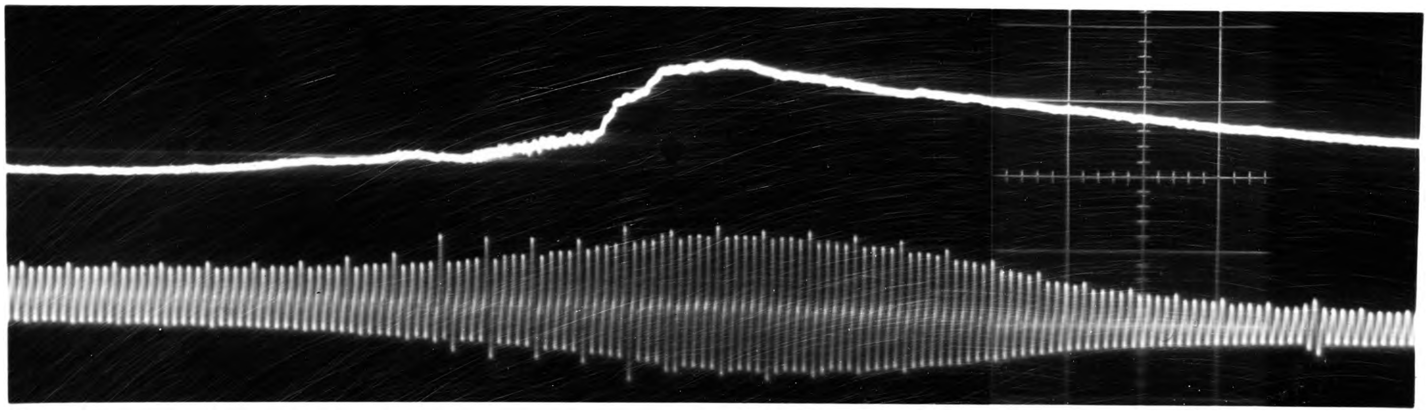


Figure 10. Temperature Versus Crank-Angle Diagram

IV. RESULTS AND DISCUSSION

Cylinder Head Temperature:

Figure 11 shows the dependence of the relative temperature, ΔTm, at a given point on the surface of the cylinder head on the fuel consumption per hour, M_f. When plotted in logarithmic coordinates the points fall in the vicinity of a straight line which represents an equation

$$\log \Delta Tm = a \log M_f + b$$

Substituting log A for b, this corresponds to the relationship

$$\Delta Tm = A M_f^a \qquad (1)$$

where

Tm = Cylinder-head surface temperature °F

T = Room temperature °F

ΔT_m = Relative cylinder-head surface temperature

= $(T_m - T)$ °F

M_f = Fuel consumption lbm/hr.

m_f = Fuel consumption lbm/cycle

n = Engine revolutions per minute

A, a = Constants

To find out value of A and a, two points on the line (Figure 11) were chosen.

$300 = A\ 5.05^a$ and

$200 = A\ 2.7^a$

Solving these two equations for the unknowns A and a,

we obtained A = 105.0 and a = 0.6483. Substituting, these values in Equation 1,

$$\Delta T_m = 105\, M_f^{0.6483}$$

Temperature factors (δ and Φ):

Differentiating Equation 1, we get

$$\delta = \frac{d\,\Delta T_m}{d\,M_f} = A\,a\,M_f^{a-1} \qquad (2)$$

or

$$\delta = \frac{a}{M_f}\,\Delta T_m$$

Substituting value of A′ and a in Equation 2

$$\delta = 68.07\, M_f^{-0.3517}$$

Equation 2 (Figure 13) shows the dependence of temperature factor (δ) on the fuel consumption per hour M_f. It shows that equal changes in M_f (in lb/hr) had a smaller influence on the relative surface temperature ΔTm at high values of M_f than at low ones.

A similar temperature factor for equal percentage increase in M_f can be obtained. From Equation 2

$$d\,\Delta T_m = \Phi = A\,a\,M_f^{a-1}\,dM_f$$

For 1% of increase in M_f, dM_f = .01 M_f, rise in temperature

$$\Phi = A\,a\,M_f^{a-1} \times 0.01\, M_f$$

$$= 0.6807\, M_f^{0.6483} \qquad (3)$$

This Φ function is plotted in Figure 12. It shows that a 1% change in M_f affects the relative surface temperature

ΔTm more stronglv at high than at low values of M_f.

Relative Temperature as Function of m_f and n:

Figure 13 shows the dependence of the relative surface temperature ΔTm on the fuel consumption per cycle, m_f for four different engine speeds n. The points in Figure 13 are more scattered than those in Figure 11 because m_f had been obtained from M_f with the aid of n, resulting in increased inaccuracy. The graphs of Figure 13 represent the relationships

$$\Delta T_m = A_1 m_f^a \qquad (4)$$

where A_1 is constant for a given n.

Figure 14 shows the dependence of the relative surface temperature on n for four different m_f. If we overlook points for 1000 rpm which is the speed beyond operating speed range of the engine, we get

$$\Delta T_m = A_2 n^a \qquad (5)$$

where A_2 is constant for a given m_f.

The exponents in Equations 4 and 5 are identical. Hence changes in ΔTm when increasing n by 1% with m_f unchanged is the same as the changes in Tm when increasing m_f by 1% with n unchanged. In both instances, m_f increases by the same 1% which is in accordance with Equation 1 and Figure 11, which also show this equivalence of n and m_f in their influence on ΔTm.

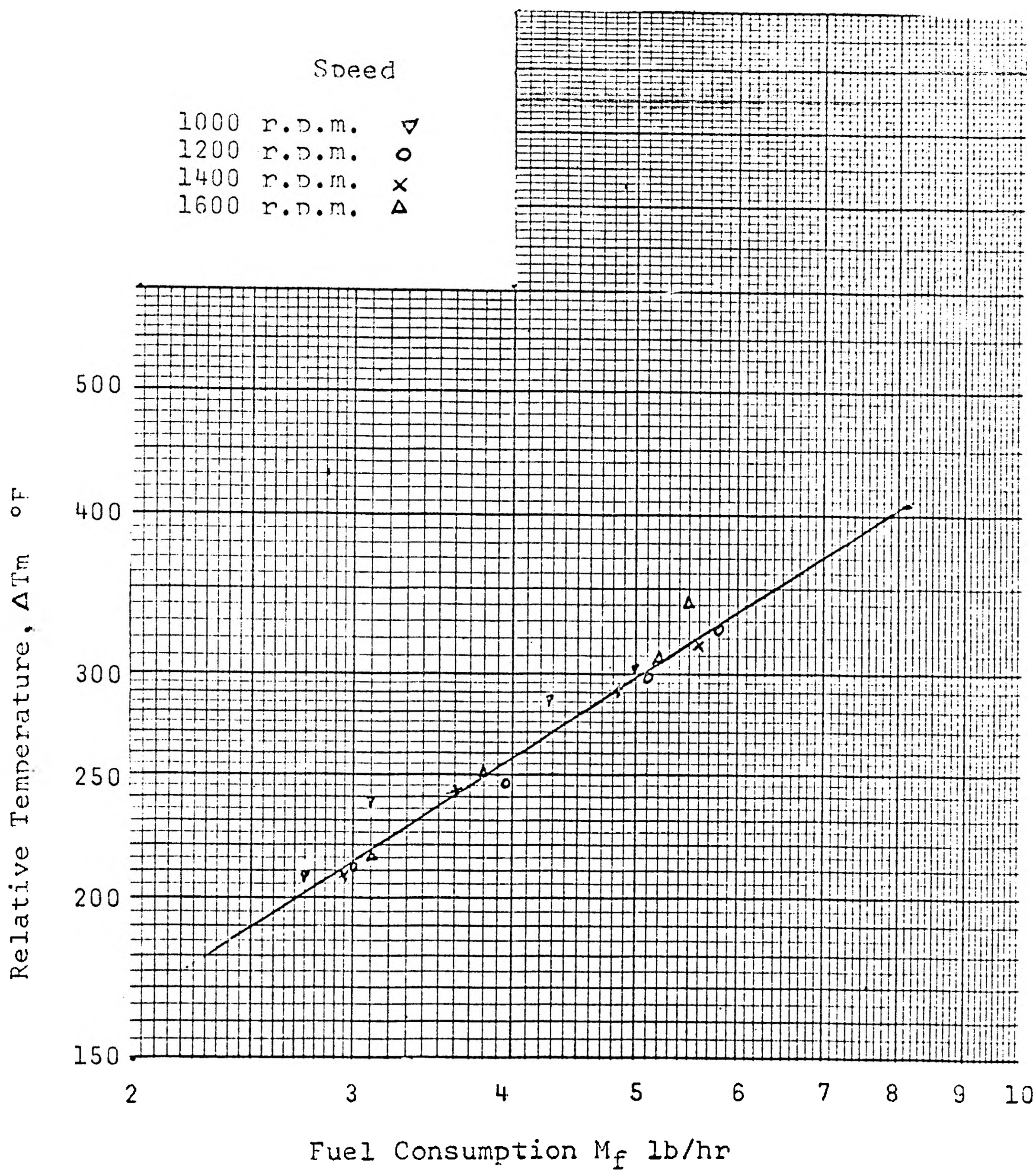
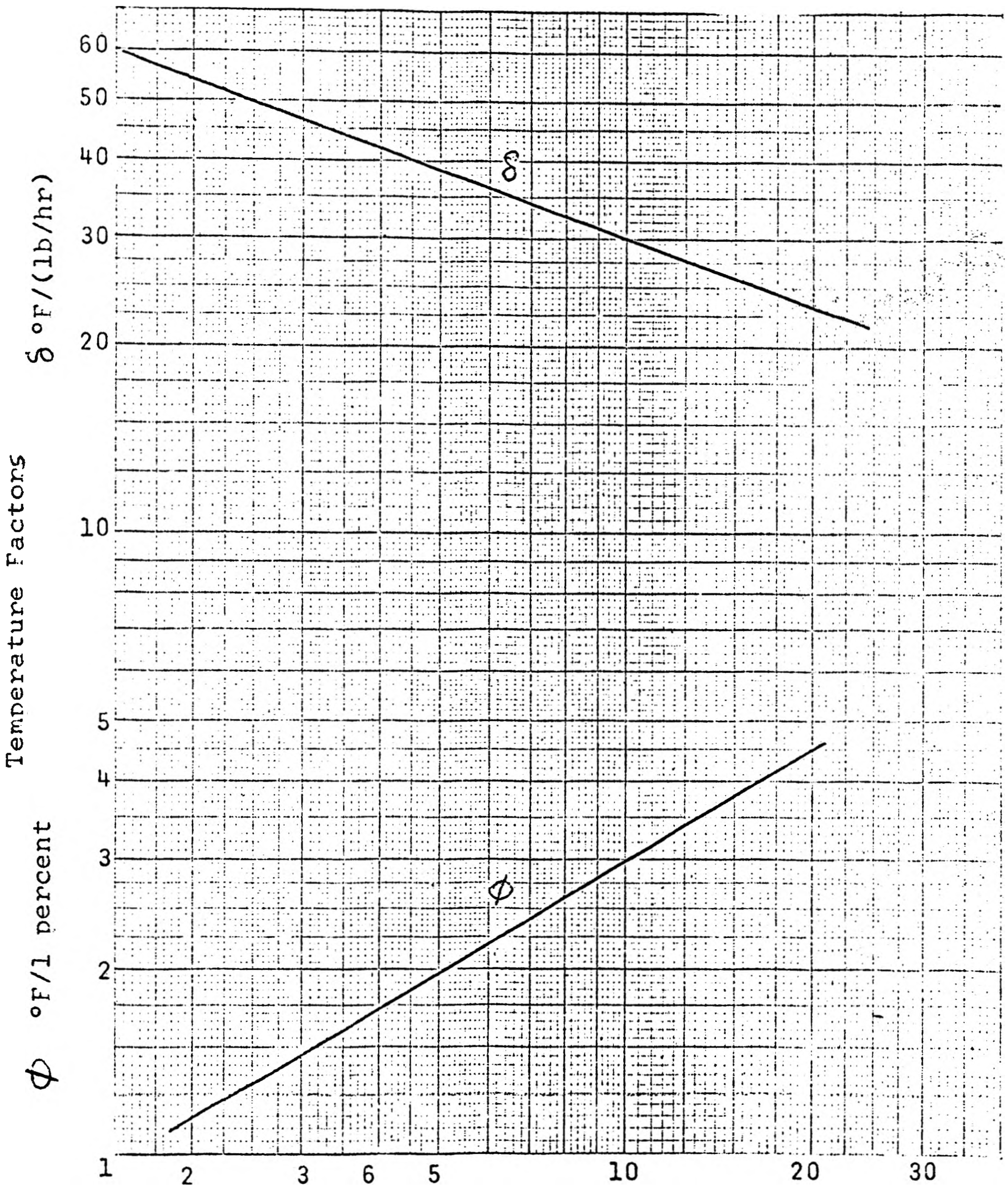


Figure 11. Relative Cylinder-Head Surface Temperature as Function of Fuel Consumption



Fuel Consumption lb/hr

Figure 12. Temperature Factor δ and Φ as Function of Fuel Consumption

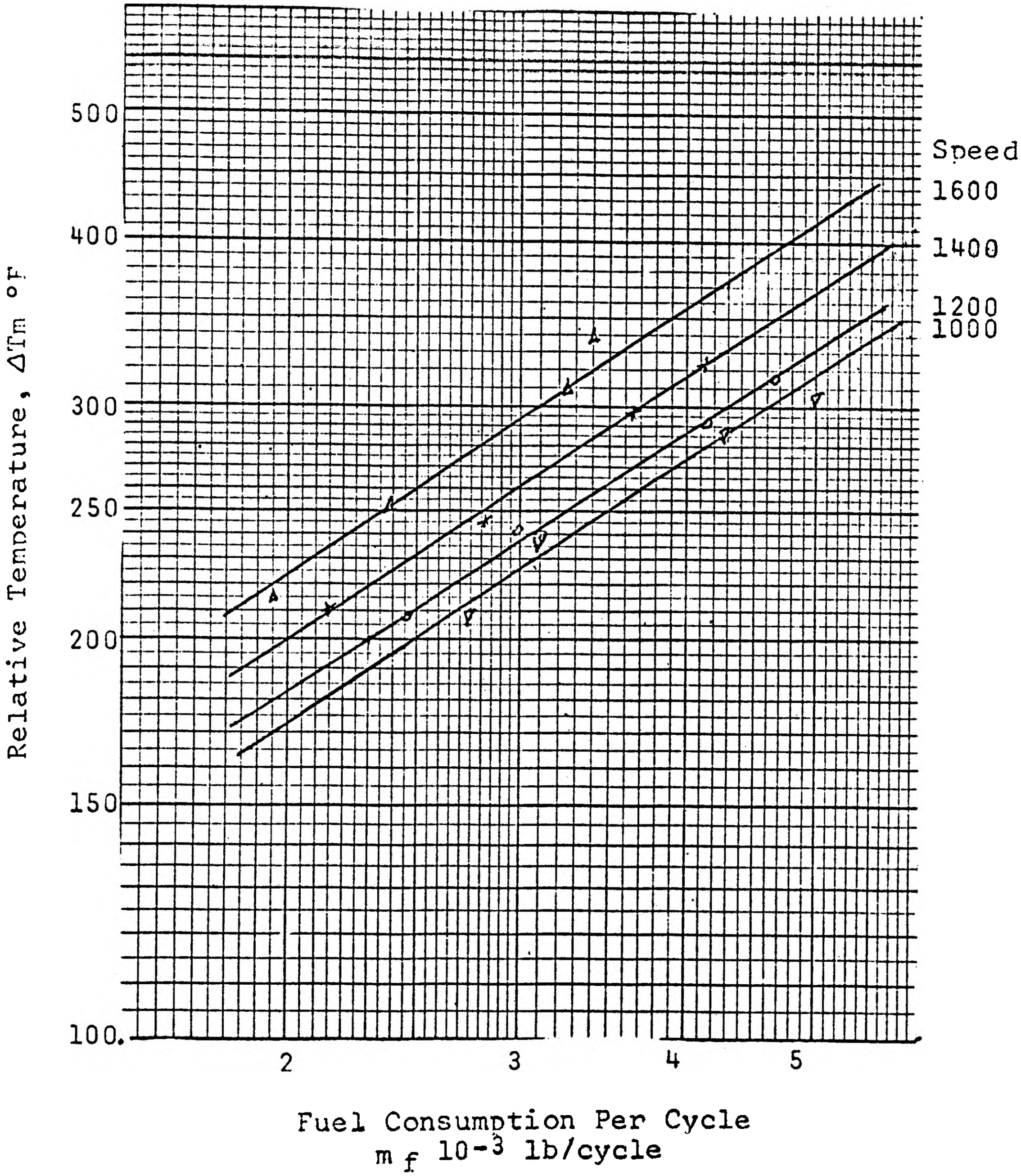


Figure 13. Relative Cylinder-Head Surface Temperature as Function of Fuel Consumption per Cycle

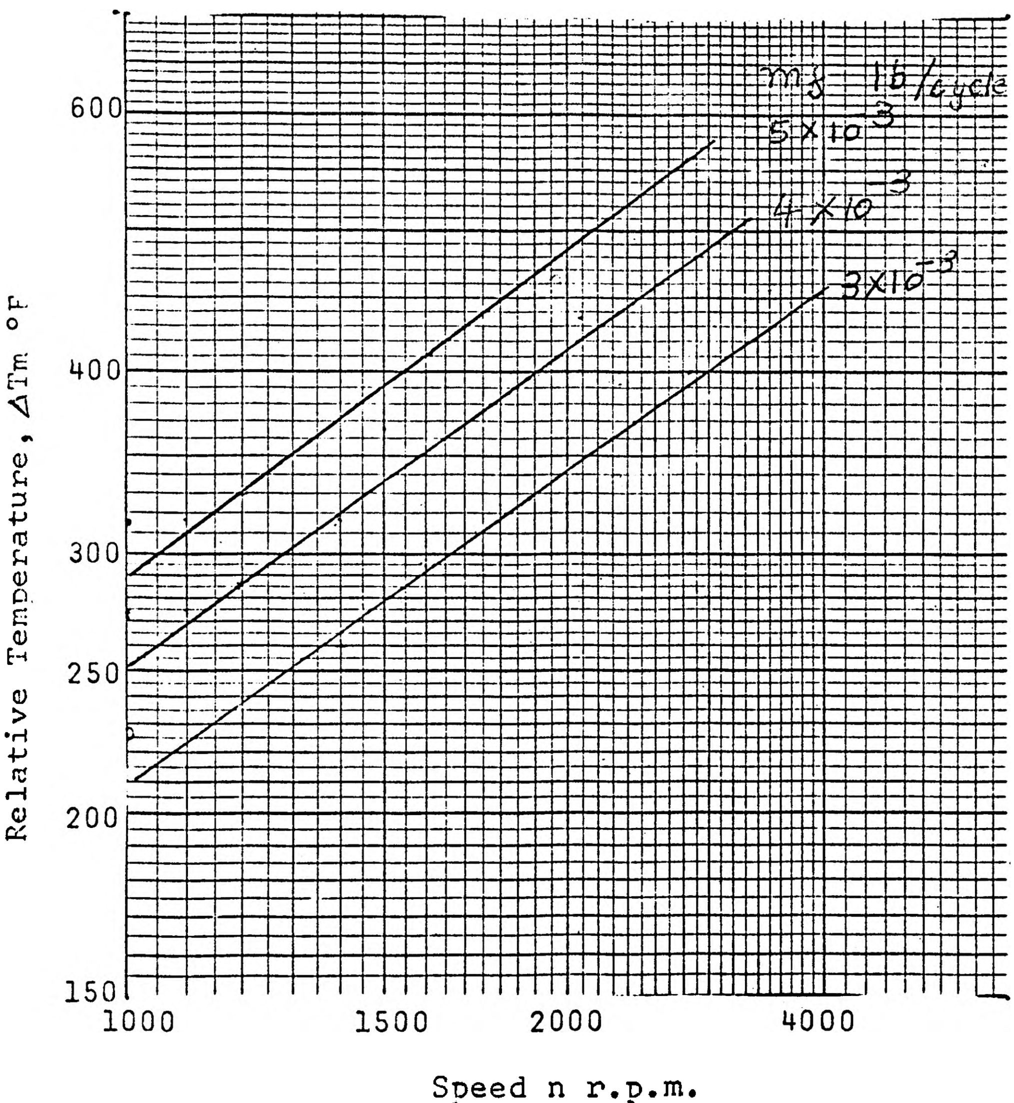


Figure 14. Relative Cylinder-Head Surface Temperature as Function of Engine Speed

Maximum Pressure:

From the pressure versus crank-angle diagrams, (Figure 16) the maximum pressures and the crank angles at the maximum pressure was tabulated in Table 2. The maximum rates of pressure-rise and corresponding crank-angles were also tabulated in the same table.

As the speed increases, the value of the maximum pressure and the maximum rate of pressure-rise both increase as may be seen in columns 3 and 5, respectively. At the same time crank angles for the maximum pressure and the maximum rate of pressure-rise also increase with speed. These can be explained in the following manner.

For a fixed ignition quality fuel, the delay period or the ignition lag depends mainly on the cylinder-air temperature. In a particular engine the cylinder air temperature at the end of compression, is almost independent of speed if other variables like outside air temperature and the compression ratio do not change. For the same delay period, delay angle increases with speed. Hence the crank-angle at the maximum pressure and maximum rate of pressure increase. Again as the delay angle increases, fuel accumulated in the cylinder during this period also increases. Hence more fuel burns in the early period of rapid combustion resulting in a higher maximum pressure and a higher maximum rate of pressure-rise.

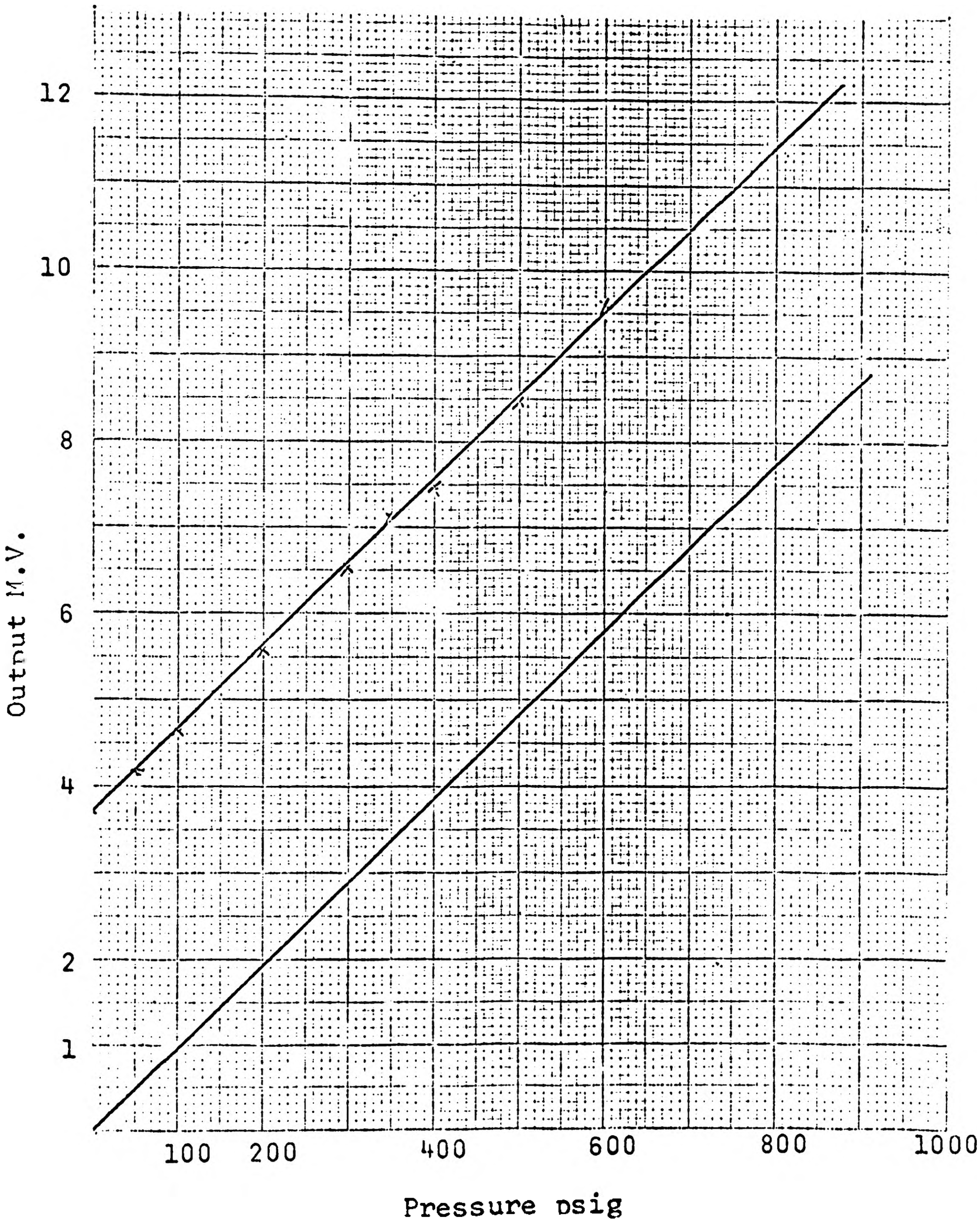


Figure 15. Calibration Curve of Pressure Transducer

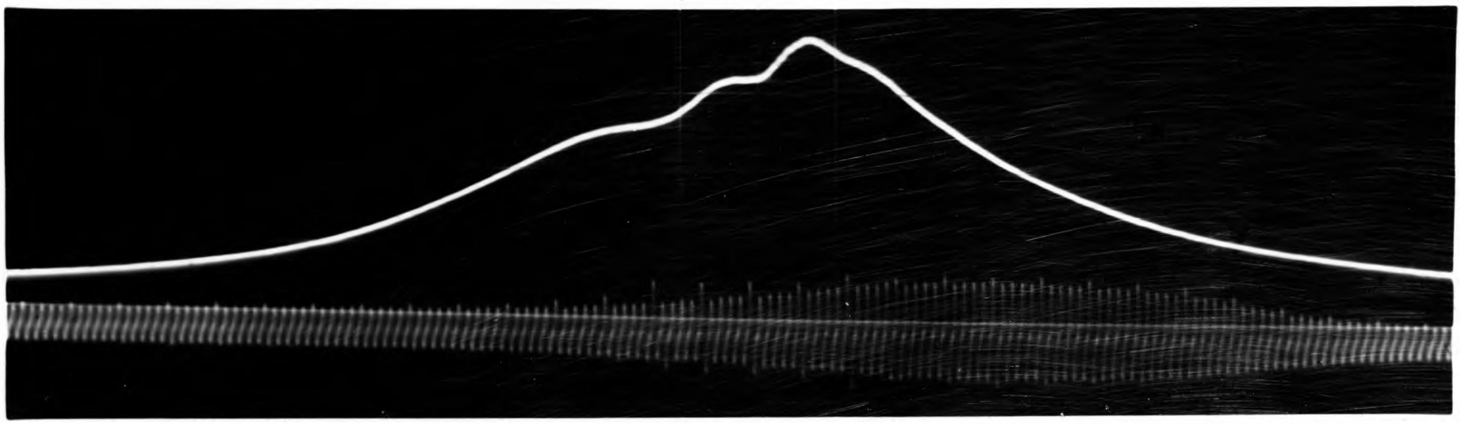


Figure 16. Pressure Versus Crank-Angle Diagram

V. CONCLUSIONS AND RECOMMENDATIONS

The results referring to surface mean temperature of the cylinder head were obtained following the reasoning developed by Gross-Gronomski (9). However, Dr. Gross-Gronomski delt with the piston temperatures and it is believed that the present investigation is the first to be similarly made with regards to the cylinder-head temperatures.

It is of interest that we have obtained a close corroboration of Dr. Gross-Gronomski's relationship between surface temperatures and the fuel consumption in diesel engines.

It is realized that many more tests with different engines and with thermocouples placed in different places are necessary before a generalization may be permissible. However, the fact that two independent researches arrived at similar conclusions dealing with different engines and different parts of the surface exposed to the working fluid, allows at least a presumption of generality.

The results referring to pressure indicate that increasing speed has caused higher rates of pressure rise and higher maximum pressure. This might cause difficulties in the development of very high-speed compression-ignition engines. For such engines, higher ignition-quality fuels should be used to keep the delay angle from becoming too great.

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APPENDIX

Table 1

Data for Calibration of Pressure Transducer

Atmospheric Pressure = 29 inches of Hg.
Temperature = 70°F.
Excitation = 12 8 V.

Input psig	Output in MV Increasing	Decreasing
0	3.731	3.554
50	4.170	4.084
100	4.646	4.566
150	5.033	5.026
200	5.516	5.536
250	5.994	6.045
300	6.499	6.520
350	7.03	7.039
400	7.436	7.521
450	7.946	8.05
500	8.451	8.52
550	8.908	8.997
600	9.446	

Table 2

Pressure Measurements

1 Load lb	2 Speed rpm	3 Maximum pressure psi	4 Angle for p max. °C.A.	5 Maximum rate of pressure rise psi/°C.A.	6 Angle for maximum rate of pressure- rise - °C. A.
20	1000	641	362	16	354-355
19.85	1200	653	364	21	355-356
20.8	1300	659	365.5	25	355-356
23.0	1400	661	367	30	356-357
21.85	1500	647	370	40	359-360
29.8	1000	659	362	14	354-355
32.2	1200	663	364.5	25	355-356
34.0	1300	700	365	30	356-357
33.0	1400	686	368	40	357-358
31.85	1500	668	369.5	44	359-360
40	1000	660	363	25	354-355
39.4	1200	680	364	30	355-356
40.8	1300	724	367	40	355-356
40.3	1400	759	369	55	358-359
42.6	1500	733	370	53.3	359-360